DUAL-SPRING COMPENSATOR ASSEMBLY FOR A FUEL INJECTOR AND METHOD

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Priority

[0001] This application claims the benefits of provisional application **S.N. 60/239,290** filed on 11 October 2000, which is hereby incorporated by reference in its entirety in this application.

Field of the Invention

[0002] The invention generally relates to a self-elongating or length-changing actuators such as an electrorestrictive, magnetorestrictive, piezoelectric or solid state actuator. In particular, the present invention relates to a compensator assembly for a length-changing actuator, and more particularly to an apparatus and method for hydraulically compensating a piezoelectrically actuated high-pressure fuel injector for internal combustion engines.

Background of the Invention

[0003] A known solid state actuator may include a ceramic structure whose axial length can change through the application of an operating voltage. It is believed that in typical applications, the axial length can change by, for example, approximately 0.12 %. In a stacked configuration, it is believed that the change in the axial length is magnified as a function of the number of actuators in the solid-state actuator stack. Because of the nature of the solid-state actuator, it is believed that a voltage application results in an instantaneous expansion of the actuator and an instantaneous movement of any structure connected to the actuator. In the field of automotive technology, especially, in internal combustion engines, it is believed that there is a need for the precise opening and closing of an injector valve element for optimizing the spray and combustion of fuel. Therefore, in

internal combustion engines, solid-state actuators are now employed for the precise opening and closing of the injector valve element.

[0004] During operation, it is believed that the components of an internal combustion engine experience significant thermal fluctuations that result in the thermal expansion or contraction of the engine components. For example, it is believed that a fuel injector assembly includes a valve body that may expand during operation due to the heat generated by the engine. Moreover, it is believed that a valve element operating within the valve body may contract due to contact with relatively cold fuel. If a solid state actuator is used for the opening and closing of an injector valve element, it is believed that the thermal fluctuations can result in valve element movements that can be characterized as an insufficient opening stroke, or an insufficient sealing stroke. It is believed that this is because of the low thermal expansion characteristics of the solid-state actuator as compared to the thermal expansion characteristics of other fuel injector or engine components. For example, it is believed that a difference in thermal expansion of the housing and actuator stack can be more than the stroke of the actuator stack. Therefore, it is believed that any contractions or expansions of a valve element can have a significant effect on fuel injector operation.

[0005] It is believed that conventional methods and apparatuses that compensate for thermal changes affecting solid state actuator operation have drawbacks in that they either only approximate the change in length, they only provide one length change compensation for the solid state actuator, or that they only accurately approximate the change in length of the solid state actuator for a narrow range of temperature changes.

[0006] It is believed that there is a need to provide thermal compensation that overcomes the drawbacks of conventional methods.

Summary of the Invention

[0007] The present invention provides a fuel injector that utilizes a length-changing actuator, such as, for example, an electrorestrictive, magnetorestrictive or a solid-state actuator with a compensator assembly that compensates for thermal distortions, brinelling, wear and mounting distortions. The compensator assembly utilizes a minimal number of elastomer seals so as to reduce a slip stick effect of such seals while achieving a more compact configuration of the compensator assembly. In one preferred embodiment of the

invention, the fuel injector comprises a housing having a first housing end and a second housing end extending along a longitudinal axis, the housing having an end member disposed between the first and second housing ends, an length-changing actuator disposed along the longitudinal axis, a closure member coupled to the length-changing solid-state actuator, the closure member being movable between a first configuration permitting fuel injection and a second configuration preventing fuel injection, and a compensator assembly that moves the solid-state actuator with respect to the body in response to temperature changes. The compensator assembly includes a body having a first body end and a second body end extending along a longitudinal axis. The body has a body inner surface facing the longitudinal axis, a first piston disposed in the body proximate one of the first body end and second body end. The first piston includes a first working surface distal to a first outer surface, the outer surface cooperating with the body inner surface to define a first fluid reservoir, a second piston disposed in the body proximate the first piston, the second piston having a second outer surface distal to a second working surface that confronts the first working surface, a first sealing member coupled to the second piston and contiguous to the body inner surface, a flexible fluid barrier coupled to the first piston and the second piston, the flexible fluid barrier cooperating with the first and second working surface to define a second fluid reservoir, and a first spring member and a second spring member. Each of the first and second spring members being contiguous to the second outer surface of the second piston so as to move at least one of the first piston and the second piston along the longitudinal axis.

[0008] The present invention provides a compensator that can be used in a length-changing actuator, such as, for example, an electrorestrictive, magnetorestrictive or a solid-state actuator so as to compensate for thermal distortion, wear, brinelling and mounting distortion of an actuator that the compensator is coupled to. In a preferred embodiment, the length-changing actuator has first and second ends. The compensator comprises a body having a first body end and a second body end extending along a longitudinal axis. The body has a body inner surface facing the longitudinal axis, a first piston disposed in the body proximate one of the first body end and second body end. The first piston includes a first working surface distal to a first outer surface, the outer surface cooperating with the body inner surface to define a first fluid reservoir, a second piston disposed in the body proximate the first piston, the second piston having a second outer surface distal to a

second working surface that confronts the first working surface, a first sealing member coupled to the second piston and contiguous to the body inner surface, a flexible fluid barrier coupled to the first piston and the second piston, the flexible fluid barrier cooperating with the first and second working surface to define a second fluid reservoir; and a first spring member and a second spring member, each of the first and second spring members being contiguous to the second outer surface of the second piston so as to move at least one of the first piston and the second piston along the longitudinal axis.

[0009] The present invention further provides a method of compensating for distortion of a fuel injector due to thermal distortion, brinelling, and wear and mounting distortion. In particular, the actuator includes a fuel injection valve or a fuel injector that incorporates a length-changing actuator such as, for example, an electrorestrictive, magnetorestrictive, piezoelectric or solid state actuator. A preferred embodiment of the length-changing actuator includes a solid-state actuator that actuates a closure member of the fuel injector. The fuel injector includes a housing having a first housing end and a second housing end extending along a longitudinal axis, the housing having an end member disposed between the first and second housing ends, an length-changing actuator disposed along the longitudinal axis, a closure member coupled to the length-changing actuator, and a compensator assembly that moves the length-changing actuator with respect to the housing in response to temperature changes. The compensator assembly includes a body having a first body end and a second body end extending along a longitudinal axis. The body has a body inner surface facing the longitudinal axis, a first piston disposed in the body proximate one of the first body end and second body end, the first piston cooperating with the body inner surface to define a first fluid reservoir, a second piston disposed in the body proximate the first piston, the second piston having a second outer surface distal to a second working surface that confronts the first working surface, an elastomer coupled to the second piston and contiguous to the body inner surface, and a flexible fluid barrier coupled to the first piston and the second piston, the flexible fluid barrier cooperating with the first and second working surface to define a second fluid reservoir. In a preferred embodiment, the method is achieved by confronting a surface of the first piston to an inner surface of the body so as to form a controlled clearance between the first piston and the body inner surface of the first fluid reservoir; engaging an elastomer between the working surface of the second piston and the inner surface of the body; coupling a flexible fluid

barrier between the first piston and the second piston such that the second piston, the elastomer and the flexible fluid barrier form the second fluid reservoir; preloading the second piston with at least one of a first spring member and a second spring member so as to generate a hydraulic pressure in the first and second hydraulic reservoirs; and biasing the length-changing actuator with a predetermined force vector resulting from changes in the volume of hydraulic fluid disposed within the first fluid reservoir as a function of temperature.

Brief Description of the Drawings

[0010] The accompanying drawings, which are incorporated herein and constitute part of this specification, illustrate presently preferred embodiments of the invention, and, together with the general description given above and the detailed description given below, serve to explain features of the invention.

[0011] Figure 1 is a cross-sectional view of a fuel injector assembly having a solid-state actuator and a compensator assembly of a preferred embodiment.

[0012] Figure 2 is an enlarged view of the compensator assembly in Figure 1.

[0013] Figure 3 is a view of the compensator of Figure 2 with a pressure sensitive valve in the first fluid reservoir.

[0014] Figure 4 is an illustration of the operation of the pressure sensitive valve of Figure 3.

Detailed Description of the Preferred Embodiments

[0015] Referring to Figures 1-4, at least one preferred embodiment is shown of a compensator assembly 200. In particular, Figure 1 illustrates a preferred embodiment of a fuel injector assembly 10 having a solid-state actuator that, preferably, includes a solid-state actuator stack 100 and a compensator assembly 200 for the stack 100. The fuel injector assembly 10 includes inlet fitting 12, injector housing 14, and valve body 17. The inlet fitting 12 includes a fuel filter 16, fuel passageways 18, 20 and 22, and a fuel inlet 24 connected to a fuel source (not shown). The inlet fitting 12 also includes an inlet end member 28 (Fig. 2) with an elastomer seal 29 that is preferably an O-ring. The inlet end member has a port 30 that can be used to fill a reservoir 32 with fluid 36 after a threaded type filler plug 38 is removed. The fluid 36 can be a substantially incompressible fluid that is responsive to temperature change by changing its volume. Preferably, the fluid 36 is

either silicon or other types of hydraulic fluid that has a higher coefficient of thermal expansion than that of the injector inlet 16, the housing 14 or other components of the fuel injector.

[0016] In the preferred embodiment, injector housing 14 encloses the solid-state actuator stack 100 and the compensator assembly 200. Valve body 17 is fixedly connected to injector housing 14 and encloses a valve closure member 40. The solid-state actuator stack 100 includes a plurality of solid-state actuators that can be operated through contact pins (not shown) that are electrically connected to a voltage source. When a voltage is applied between the contact pins (not shown), the solid-state actuator stack 100 expands in a lengthwise direction. A typical expansion of the solid-state actuator stack 100 may be on the order of approximately 30-50 microns, for example. The lengthwise expansion can be utilized for operating the injection valve closure member 40 for the fuel injector assembly 10. That is, the lengthwise expansion of the stack 100 and the closure member 40 can be used to define an orifice size of the fuel injector as opposed to an orifice of a valve seat or an orifice plate as is used in a conventional fuel injector.

[0017] Solid-state actuator stack 100 is guided along housing 14 by means of guides 110. The solid-state actuator stack 100 has a first end in operative contact with a closure end 42 of the valve closure member 40 by means of bottom 44, and a second end of the stack 100 that is operatively connected to compensator assembly 200 by means of a top 46.

[0018] Fuel injector assembly 10 further includes a spring 48, a spring washer 50, a keeper 52, a bushing 54, a valve closure member seat 56, a bellows 58, and an O-ring 60. O-ring 60 is preferably a fuel compatible O-ring that remains operational at low ambient temperatures (-40 Celsius° or less) and at operating temperatures (140 Celsius° or more).

[0019] Referring to Fig. 2, compensator assembly 200 includes a body 210 encasing a first piston 220, a piston stem or an extension portion 230, a second piston 240, bellows 250 and elastic member or first spring 260. The body 210 can be of any suitable cross-sectional shape as long as it provides a mating fit with the first and second pistons, such as, for example, oval, square, rectangular or any suitable polygons. Preferably, the cross section of the body 210 is circular, thereby forming a cylindrical body that extends along the longitudinal axis A-A.

[0020] The extension portion 230 extends from the first piston 220 so as to be linked by an extension end 232 to the top 46 of the piezoelectric stack 100. Preferably, the

extension portion 230 is integrally formed as a single piece with the first piston 220. Alternatively, the extension portion can be formed as a separate piece from the first piston 220, and coupled to the first piston 220 by, for example, a spline coupling, ball joint, a heim joint or other suitable couplings that allow two moving parts to be coupled together.

[0021] First piston 220 is disposed in a confronting arrangement with the inlet end member 28. An outer peripheral surface 228 of the first piston 220 is dimensioned so as to form a close tolerance fit with a body inner surface 212, i.e. a controlled clearance that allows lubrication of the piston and the body while also forming a hydraulic seal that controls the amount of fluid leakage through the clearance. The controlled clearance between the first piston 220 and body 210 provides a controlled leakage flow path from the first fluid reservoir 32 to the second fluid reservoir 33, and reduces friction between the first piston 220 and the body 210, thereby minimizing hysteresis in the movement of the first piston 220. It is believed that side loads introduced by the stack 100 would increase the friction and hysteresis. As such, the first piston 220 is coupled to the stack 100 preferably only in a direction along the longitudinal axis A-A so as to reduce or even eliminate any side loads. The body 210 is free floating relative to the injector housing, thus operate to reduce or even prevent distortion of the injector housing. Furthermore, by having a spring contained within the piston subassembly, little or no external side forces or moments are introduced by the compensator assembly 200 to the injector housing.

[0022] To permit fluid 36 to selectively circulate between a first face 222 of the first piston 220 and a second face 224 of the first piston 220, a passage 226 extends between the first and second faces. Pockets or channels 228a can be formed on the first face 222 that are in fluid communication with the second fluid reservoir 33 via the passage 226. The pockets 228a ensure that some fluid 36 can remain on the first face 222 to act as a hydraulic "shim" even when there is little or no fluid between the first face 222 and the end member 28. In a preferred embodiment, the first reservoir 32 always has at least some fluid disposed therein. The first face 222 and the second face 224 can be of any shapes such as, for example, a conic surface of revolution, a frustoconical surface or a planar surface. Preferably, the first face 222 and second face 224 include a planar surface transverse to the longitudinal axis A-A.

[0023] Disposed between the first piston 220 and the top 46 of the stack 100 is a ring like piston or second piston 240 mounted on the extension portion 230 so as to be axially

slidable along the longitudinal axis A-A. The second piston 240 includes a sealing member, preferably an elastomer 242 disposed in a groove 245 on the outer circumference of the second piston 240 so as to generally prevent leakage of fluid 36 towards the stack 100. Preferably, the elastomer 242 is an O-ring. Alternatively, the elastomer 242 can be an O-ring of the type having non-circular cross-sections. Other types of elastomer seal can also be used, such as, for example, a labyrinth seal.

[0024] The second piston includes a surface 246 that forms, in conjunction with a surface 256 of the first bellows collar 252, a second working surface 248. Here, the second working surface is disposed in a confronting arrangement with the first working surface, i.e. the second face 224 of the first piston 220. Preferably, the pistons are circular in shape, although other shapes, such as rectangular or oval, can also be used for the piston 220.

[0025] The second piston 240 is coupled to the extension portion 230 via bellows 250 and at least one elastic member, preferably a first spring 260 and a second spring 262. The first spring 260 is confined between a first boss portion 280 of the extension portion 230 and the second piston 240. The second spring 262 is confined between the second piston 240 and a second boss portion 282 that is coupled to the body 210. Preferably, the first boss portion 280 can be a spring washer that is affixed to the extension portion by a suitable technique, such as, for example, threading, welding, bonding, brazing, gluing and preferably laser welding. The bellows 250 includes a first bellows collar 252 and a second bellows collar 254. The first bellows collar 252 is affixed to the inner surface 244 of the second piston 240. The second bellows collar 254 is affixed to the first boss portion 280. Both of the bellows collars can be affixed by a suitable technique, such as, for example, threading, welding, bonding, brazing, gluing and preferably laser welding. It should be noted here that the first bellows collar 252 is disposed for a sliding fit on the extension portion 230. Preferably, the first bellows collar 252 in its axial neutral (unloaded) condition has approximately 300 micrometer of clearance between the extension portion 230 and the bellows collar 252 at room temperature (approximately 20 degrees Celsius). From this position the clearance can change between approximately +/- 100 microns to approximately +/- 300 microns depending on the number of operating cycles that are desired for the solid state actuator. Maximum operating temperature (approximately 140 degrees Celsius or greater) could increase this clearance to approximately 400 microns.

Minimum operating temperature (approximately -40 degrees Celsius or lower) would decrease the clearance to approximately 250 microns.

[0026] The first spring 260 and the second spring 262 can react against their respective boss portions 280, 282 to push the second working surface 248 towards the inlet 16. This causes a pressure increase in the fluid 36 that acts against the first face 222 and second face 224 of the first piston 220. In an initial condition, hydraulic fluid 36 is pressurized as a function of the product of the combined spring force of the first and second springs and the surface area of the second working surface 248. Prior to any expansion of the fluid in the first reservoir 32, the first reservoir is preloaded so as to form a hydraulic shim. Preferably, each of the spring force of first spring 260 or the second spring 262 is approximately 30 Newton to 70 Newton.

[0027] The fluid 36 in the first fluid reservoir 32 that forms a hydraulic shim tends to expand due to an increase in temperature in and around the compensator assembly 200. Since the first face 222 has a greater surface area than the second working surface 248, the first piston 220 tends to move towards the stack or valve closure member 40. The force vector (i.e. having a direction and magnitude) "F_{out}" of the first piston 220 moving towards the stack is defined as follows:

[0028] $F_{out} = F_{spring262} - [(F_{spring260} + F_{spring262} \quad F_{seal})*((A_{shim} / A_{2ndReservoir})-1)]$

[0029] where:

F_{out} = Applied Force (To the Piezo Stack)

$F_{spring260}$ = Spring Force of Spring 260

$F_{spring262}$ = Spring Force of Spring 262

F_{seal} = Seal Friction Force (sealing member 242)

$A_{shim} = (\pi/4) * Pd^2$ or Area above piston where Pd is first piston diameter

$A_{2ndReservoir} = (\pi/4) * (Pd^2 - Bh^2)$ or Area below the first piston where Bh is the hydraulic diameter of bellows 250

[0030] At rest, the respective pressure of the pressures in the hydraulic shim and the second fluid reservoir tends to be generally equal. Since the friction force of sealing member 242 affects the pressure in the hydraulic shim and the second fluid reservoir equally, the sealing member 242 does not affect the force F_{out} of the piston. However, when the solid-state actuator is energized, the pressure in the hydraulic shim is generally increased because of the relatively large combined spring force (of the springs 260 and 262) as the stack expands. This allows the stack 100 to have a relatively stiff reaction base in which the valve closure member 40 can be actuated so as to inject fuel through the fuel outlet 62.

[0031] Preferably, each of the first spring 260 and the second spring 262 is a coil spring. Here, the pressure in the fluid reservoirs is related to at least one spring characteristic of each of the coil springs. As used throughout this disclosure, the at least one spring characteristic can include, for example, the spring constant, spring free length and modulus of elasticity of the spring. Each of the spring characteristics can be selected in various combinations with other spring characteristic(s) noted above so as to achieve a desired response of the compensator assembly. Furthermore, due to the use of at least two springs, the compensator is under a relatively high pressure (10 to 15 bars) operating range which range is believed to reduce the need for a high vacuum (so as to reduce the amount of dissolved gases) during a filling of the compensator assembly 200, and also the need for a pressure responsive valve that would be needed to isolate the first fluid reservoir 32 from the second fluid reservoir during an activation of the actuator stack 100.

[0032] However, it is also preferable to include a valve to prevent hydraulic fluid from flowing out of the first reservoir 32 as a function of the pressure in the first or second fluid reservoirs. The valve can include, for example, a pressure responsive valve, a check valve or a one-way valve. Preferably, the valve is a plate type valve, referenced as numeral 270 in Figure 3. Specifically, the pressure sensitive valve is a flexible thin-disc plate 270 having a smooth surface disposed atop the first face 222 as shown in Figure 4.

[0033] In particular, by having a smooth surface on the side contiguous to the first piston 220 that forms a sealing surface 274 with the first face 222, the plate 270 functions as a pressure sensitive valve that allows fluid to flow between a first fluid reservoir 32 and a second fluid reservoir 33 whenever pressure in the first fluid reservoir 32 is less than pressure in the second reservoir 33. That is, whenever there is a pressure differential

between the reservoirs, the smooth surface of the plate 270 is lifted up to allow fluid to flow to the channels or pockets 228a. It should be noted here that the plate forms a seal to prevent flow as a function of the pressure differential instead of a combination of fluid pressure and spring force as in a ball type check valve. The pressure sensitive valve or plate 270 includes at least one orifice 272 formed through its surface. The orifice can be, for example, square, circular or any suitable through orifice. Preferably, there are twelve orifices formed in the plate. The plate 270 is preferably welded to the first face 222 at four or more different points 276 around the perimeter of the plate 270.

[0034] Because the plate 270 has very low mass and is flexible, it responds very quickly with the incoming fluid by lifting up towards the end member 28 so that fluid that has not passed through the plate adds to the volume of the hydraulic shim. The plate 270 approximates a portion of a spherical shape as it pulls in a volume of fluid that is still under the plate 270 and in the passage 226. This additional volume is then added to the shim volume but whose additional volume is still on the first reservoir side of the sealing surface. One of the many benefits of the plate 270 is that pressure pulsations are quickly damped by the additional volume of hydraulic fluid that is added to the hydraulic shim in the first reservoir. This is because activation of the injector is a very dynamic event and the transition between inactive, active and inactive creates inertia forces that produce pressure fluctuations in the hydraulic shim. The hydraulic shim, because it has free flow in and restricted flow out of the hydraulic fluid, quickly dampens the oscillations.

[0035] The through hole or orifice diameter of the at least one orifice 272 can be thought of as the effective orifice diameter of the plate instead of the lift height of the plate 270 because the plate 270 approximates a portion of a spherical shape as it lifts away from the first face 222. Moreover, the number of orifices and the diameter of each orifice determine the stiffness of the plate 270, which is critical to a determination of the pressure drop across the plate 270. Preferably, the pressure drop should be small as compared to the pressure pulsations in the first reservoir 32 of the compensator. When the plate 270 has lifted approximately 0.1 mm, the plate 270 can be assumed to be wide open, thereby giving unrestricted flow into the first reservoir 32. The ability to allow unrestricted flow into the hydraulic shim prevents a significant pressure drop in the fluid. This is important because when there is a significant pressure drop, the gas dissolved in the fluid comes out, forming bubbles. This is due to the vapor pressure of the gas exceeding the reduced fluid pressure

(i.e. certain types of fluid take on air like a sponge takes on water, thus, making the fluid behave like a compressible fluid.) The bubbles formed act like little springs making the compensator "soft" or "spongy". Once formed, it is difficult for these bubbles to re-dissolve into the fluid. The compensator, preferably by design, operates between approximately 10 to 15 bars of pressure and it is believed that the hydraulic shim pressure does not drop significantly below atmospheric pressure. Thus, degassing of the fluid and compensator passages is not as critical as it would be without the plate 270. Preferably, the thickness of the plate 270 is approximately 0.1 millimeter and its surface area is approximately 110 millimeter squared. Furthermore, to maintain a desired flexibility of the plate 270, it is preferable to have an array of approximately twelve orifices, each orifice having an opening of approximately 0.8 millimeter squared (mm^2), and the thickness of the plate is preferably the result of the square root of the surface area divided by approximately 94.

[0036] Referring again to Figure 1, during operation of the fuel injector 10, fuel is introduced at fuel inlet 24 from a fuel supply (not shown). Fuel at fuel inlet 24 passes through a fuel filter 16, through a passageway 18, through a passageway 20, through a fuel tube 22, and out through a fuel outlet 62 when valve closure member 40 is moved to an open configuration.

[0037] In order for fuel to exit through fuel outlet 62, voltage is supplied to solid-state actuator stack 100, causing it to expand. The expansion of solid-state actuator stack 100 causes bottom 44 to push against valve closure member 40, allowing fuel to exit the fuel outlet 62. After fuel is injected through fuel outlet 62, the voltage supply to solid-state actuator stack 100 is terminated and valve closure member 40 is returned under the bias of spring 48 to close fuel outlet 62. Specifically, the solid-state actuator stack 100 contracts when the voltage supply is terminated, and the bias of the spring 48 which holds the valve closure member 40 in constant contact with bottom 44, also biases the valve closure member 40 to the closed configuration.

[0038] In the preferred embodiment of Figure 3, when the actuator 100 is energized, pressure in the first reservoir 32 increases rapidly, causing the plate 270 to seal tight against the first face 222. This blocks the hydraulic fluid 36 from flowing out of the first fluid reservoir to the passage 236. It should be noted that the volume of the shim during activation of the stack 100 is related to the volume of the hydraulic fluid in the first

reservoir at the approximate instant the actuator 100 is activated. Because of the virtual incompressibility of fluid, the fluid 36 in the first reservoir 32 approximates a stiff reaction base, i.e. a shim, on which the actuator 100 can react against. The stiffness of the shim is believed to be due in part to the virtual incompressibility of the fluid and the blockage of flow out of the first reservoir 32 by the plate 270. Here, when the actuator stack 100 is actuated in an unloaded condition, it extends by approximately 60 microns. As installed in a preferred embodiment, one-half of the quantity of extension (approximately 30 microns) is absorbed by various components in the fuel injector. The remaining one-half of the total extension of the stack 100 (approximately 30 microns) is used to deflect the closure member 40. Thus, a deflection of the actuator stack 100 is constant, as it is energized time after time, thereby allowing an opening of the fuel injector to remain the same.

[0039] Referring to Figure 1, as valve closure member 40 contracts, bottom 44 of the actuator stack 100 tends to separate from its contact point with valve closure end 42. Length-changing actuator stack 100, which is operatively connected to the bottom surface of first piston 220, is initially pushed downward due to a pressurization of the fluid by the springs 260, 262 acting on the second piston with a force F_{out}. The increase in temperature causes inlet fitting 12, injector housing 14 and valve body 17 to expand relative to the actuator stack 100 due to the generally higher volumetric thermal expansion coefficient β of the fuel injector components relative to that of the actuator stack. This movement of the first piston is transmitted to the actuator stack 100 by a top 46, which movement maintains the position of the bottom 44 of the stack constant relative to the closure end 42 of the closure member 40. It should be noted that in the preferred embodiments, the thermal coefficient β of the hydraulic fluid 36 is greater than the thermal coefficient β of the actuator stack. Here, the compensator assembly can be configured by at least selecting a hydraulic fluid with a desired coefficient β and selecting a predetermined volume of fluid in the first reservoir such that a difference in the expansion rate of the housing of the fuel injector and the actuator stack 100 can be compensated by the expansion of the hydraulic fluid 36 in the first reservoir.

[0040] In the preferred embodiment of Fig. 2, when the actuator 100 is energized, pressure in the first reservoir 32 increases rapidly due in part to the high operating pressure in the compensator. Because of the high operating pressure and virtual incompressibility of fluid, the fluid 36 in the first reservoir 32 approximates a stiff reaction base, i.e. a shim, on which

the actuator 100 can react against. Here, when the actuator stack 100 is actuated in an unloaded condition, it extends by approximately 60 microns. As installed in a preferred embodiment, one-half of the quantity of extension (approximately 30 microns) is absorbed by various components in the fuel injector. The remaining one-half of the total extension of the stack 100 (approximately 30 microns) is used to deflect the closure member 40. Thus, a deflection of the actuator stack 100 is believed to be constant, as it is energized time after time, thereby allowing an opening of the fuel injector to remain the same.

[0041] When the actuator 100 is not energized, fluid 36 flows between the first fluid reservoir and the second fluid reservoir while maintaining the same preload force F_{out}. The force F_{out} is a function of the springs 260, 262, the friction force due to the seal 242 and the surface area of each piston. Thus, it is believed that the bottom 44 of the actuator stack 100 is maintained in constant contact with the contact surface of valve closure end 42 regardless of expansion or contraction of the fuel injector components.

[0042] Although the compensator assembly 200 has been shown in combination with a piezoelectric actuator for a fuel injector, it should be understood that any length changing actuator, such as, for example, an electrorestrictive, magnetorestrictive or a solid-state actuator could be used with the compensator assembly 200. Here, the length changing actuator can also involve a normally deenergized actuator whose length is expanded when the actuator energized. Conversely, the length-changing actuator is also applicable to where the actuator is normally energized and is de-energized so as to cause a contraction (instead of an expansion) in length. Moreover, it should be emphasized that the compensator assembly 200 and the length-changing solid state actuator are not limited to applications involving fuel injectors, but can be for other applications requiring a suitably precise actuator, such as, to name a few, switches, optical read/write actuator or medical fluid delivery devices.

[0043] While the present invention has been disclosed with reference to certain preferred embodiments, numerous modifications, alterations, and changes to the described embodiments are possible without departing from the sphere and scope of the present invention, as defined in the appended claims. Accordingly, it is intended that the present invention not be limited to the described embodiments, but that it have the full scope defined by the language of the following claims, and equivalents thereof.